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Eugene R. Montalvo
Date: April 14, 2004

IN THE UNITED STATES PATENT AND TRADEMARK OFFICE

In re application of	
Donald W. Allen et al	
Serial No. 09/625,893	Group Art Unit: 3677
Filed: July 26, 2000	Examiner: K. Mitchell
SMOOTH SLEEVES FOR DRAG AND VIV REDUCTION OF CYLINDRICAL STRUCTURES	April 14, 2004

Mail Stop Appeal Brief – Patents
Commissioner for Patents
P.O. Box 1450
Alexandria, Virginia 22313-1450

Sir:

APPELLANT SHELL OIL COMPANY'S BRIEF

Attorney for Assignee in the above captioned application, hereby files Appellant Shell Oil Company's Brief, following the filing of a Notice of Appeal on 24 February 2004. Shell Oil Company is assignee of the claimed invention pursuant to a Notice of Assignment recorded at Reel 014182 / Frame 0244 on 18 June 2003.

TABLE OF AUTHORITIES

Cases

Statutes

Other Authorities

CERTIFICATE OF INTEREST

The real party at interest in the present appeal is assignee, Shell Oil Company.

STATEMENT OF RELATED APPEALS

There are no other appeals or interferences known to appellant, appellant's counsel, or assignee which will directly affect or be directly affected by or have a bearing on the Board's decision in the present appeal.

STATUS OF THE CLAIMS

Claims 1 and 4 stand rejected under 35 U.S.C. §103(a) as being obvious over Allen, D, Henning, D. *Vortex Induced Vibration Tests of a Flexible Smooth Cylinder at Supercritical Reynolds Numbers*, Proceedings of the Seventh International Offshore and Polar Engineering Conference, Honolulu, Hawaii, U.S.A., May 1997 (the "Allen Paper"). Claims 2, 3, 5 and 6 stand rejected under 35 U.S.C. §103(a) as being obvious over the Allen Paper in view of U.S. Patent 4,470,722 to Gregory et al. ("Gregory '722"). Claim 3 further stands objected to in that an amendment made to the claim on in a 19 August 2003 Response to the prior Office Action failed to indicate that the claim was amended. Claims as they presently stand are attached as Appendix A hereto.

STATUS OF AMENDMENTS

No amendments are pending following the issuance of the Final Office Action dated 2 December 2003.

SUMMARY OF THE INVENTION

The claimed invention is directed to a method and apparatus for reducing drag and vortex-induced vibration (VIV) seen by cylindrical elements in a marine environment. Drag and VIV are two distinct phenomenon seen by bodies in a moving fluid, such as ocean currents. Drag is the force upon the body caused by the moving fluid. VIV, on the other hand, is a phenomenon that is caused by fluid vortexes shedding from the body in an alternate fashion causing the body to undergo cyclical vibration. Both can result in stress and fatigue in bodies in a moving fluid. Spec. page 1, line 21 – page 2, line 3. Existing solutions to VIV such as helical strakes on the body may increase drag. Other VIV solutions such as fairings tend to be expensive and difficult to install. Thus, there exists a need for a solution that addresses both drag and VIV.

The present invention claims the use of an ultra-smooth surface for a body to reduce both drag and VIV. It has been found to be particularly effective in certain Reynolds number ranges. The Reynolds number is a common fluid dynamics term and can be expressed as :

$$R_e = \frac{Dv\rho}{\mu}$$

where D is the diameter of the body; v, the velocity of the moving fluid; ρ, the density of the fluid; and μ, the dynamic viscosity of the fluid. The term ultra-smooth surface is defined in the specification as a surface having a K/D ratio of $10\text{x}10^{-4}$ or less, where K is the average measured peak to trough distance on the surface and D is the effective outside diameter of the cylindrical element. The claims include K/D surface ratios achieved by preparation of the surface of the body, a coating on the surface of the body or the placement of substantially cylindrical sleeves having the required K/D ratio about the body. Spec. page 7, line 4 – page 8 line 6.

Examples of how to achieve providing a body with an effective ultrasmooth surface are show in Figs. 1 – 4. Where there are multiple cylindrical bodies grouped together, such as risers and umbilicals, a shroud or sleeve has been designed to

accommodate multiple elements. See, Figs. 6 – 8. The sleeve may further be part of a buoyancy module.

Figs. 15 and 16 demonstrate that the RMS displacement of the body having a smooth surface decreases in comparison to non-ultra-smooth surfaces and drop significantly where the Reynolds number is in the super critical range. Thus, the present invention provides for a relatively low cost method for addressing both drag and VIV.

STATEMENT OF THE ISSUES

1. The Examiner erred in rejecting claims 1 and 4 as being obvious over the Allen Paper, asserting that the Allen Paper inherently disclosed the present invention, particularly in light of the Declaration of Dr. Allen (paper co-author) submitted in a Response filed 27 August 2002.

2. The Examiner erred in rejecting claims 2, 3, 5, and 6 as being obvious over the Allen Paper, as is relevant to issue 1, in combination with the Gregory '722 reference.

GROUPING OF THE CLAIMS

As noted above in the STATUS OF THE CLAIMS section, claims 1 and 4 are rejected solely on the Allen Paper. However, since claims 1 and 4 are both independent claims, and it is not believed that they fall or stand together. Claims 2, 3, 5 and 6 are rejected on the Allen Paper and Gregory '722. Again, each claim is independent and it is believed that the claims would not stand or fall together.

ARGUMENT

I. THE EXAMINER ERRED IN FINDING CLAIMS 1 AND 4 OBVIOUS UNDER 35 U.S.C. §103(a) OVER THE ALLEN PAPER

The Examiner, in the Final Office Action mailed 2 December 2003, rejected claims 1 and 4 under 35 U.S.C. §103(a) as being obvious over the Allen Paper. Specifically, it was argued that the Allen Paper taught a method and system for controlling drag and vortex induced vibration, consisting of providing an ultra-smooth surface about the cylinder element of ABS® or PVC plastic with a surface roughness of k/D between 8.86×10^{-5} to 1.51×10^{-4}, citing Allen Paper page 681, col. 1, 2nd – 4th full paragraph. A copy of the Allen Paper is attached as Appendix B hereto. The Examiner further noted that on page 683 of the Allen Article that "Stationary Cylinder Results" stated that the differences between the data cited therein and data from the resources in Fig. 4 was probably related to surface roughness, as is consistent with certain references, including Shih et al. (1992). The Examiner went on to state that Page 684 of the Allen Paper concludes that the tests using the cylinder elements of said k/D range resulted in determining that surface roughness had an important effect on drag and VIV response of circular cylinders and the apparatus was inherently taught by the method. Final Office Action of 2 December 2003, ¶3, page 3.

A. The Teachings of the Allen Paper

The Allen Paper dealt with a series of test designed to determine the effects of drag and VIV on smooth cylinders at supercritical Reynolds numbers. The range of Reynolds numbers were defined, for the purposes of the paper, at Page 683, col. 1, Test Results, second paragraph:

below critical	$< 1 \times 10^{5}$
critical	$1 \times 10^{5} - 4 \times 10^{5}$
supercritical	$4 \times 10^{5} - 3.5 \times 10^{6}$
transcritical	$> 3.5 \times 10^{6}$

Multiple cylinders of two different types, a 145 psi ABS pipe and a schedule 40 PVC pipe were used in the test. The cylinder information set forth at page 681, col. 1, Test Setup, is summarized in Table 1 below:

Type	O.D.	Wall Thk.	Mod. Elasticity	Range of K/D	Nat. ω Calc.	Nat. ω Meas.
ABS	3.5 in	0.247 in.	220 ksi	8.86×10^{-5} – 1.09×10^{-4}	5.77 Hz	N/A
PVC	5.5625 in.	0.258 in.	457 ksi	1.21×10^{-4} – 1.51×10^{-4}	8.83 Hz	8.48 Hz

Table 1

The Allen Paper goes on to discuss the test setup at page 681, column 2. As it notes, for each pipe size, there were two pipe holders fabricated. The first was for the pipe in which an aluminum pipe, or "strongback" was inserted inside the ABS or PVC cylinder; the second was for the pipe size without a strongback. As noted in the last paragraph of page 681, column 2:

> A number of tests were conducted using a "strongback" mounted insider the test cylinder. . . . The function of the strongback was to restrain the test cylinder from experiencing vortex-induced vibrations so that stationary cylinder drag could be measured. The presence of the strongback accomplished this by increasing the natural frequency in bending to a value that is higher than expected to be excited during the tests.

The Allen Paper goes on to describe the specific instrumentation and the flow regimes under which the pipes were tested at pages 682 – 683.

Figure 4 is a plot of the drag coefficients, with other data from researchers for the tests <u>with a strongback</u>. Fig. 4 noted a steep decrease in the drag coefficient through the critical Reynolds number range, known as the "drag crisis" and continues to slightly decrease as the Reynolds number continues to decrease. All of the data, including the experimental data then shows an increase in drag coefficient as the Reynolds number approaches the transcritical range.

Figure 5 is a plot of the results for the smooth ABS cylinder without a strongback, thus making the cylinder free to experience VIV. As the paper notes at page 683, col. 2:

> The substantial VIV resulted in a significant increase in the drag coefficients (about 50 – 80 percent). These results illustrate that significant VIV can, and does, occur at critical and supercritical Reynolds numbers and that the corresponding increase in drag coefficient is similar to that of subcritical Reynolds numbers.

Figure 6 is plot of the results for the smooth PVC cylinder without a strongback, and shows substantial vibration and significant increased drag due to vibrations. As the Allen Paper notes, the fluid velocities were not increased until the pipe "locked out" because the pipe could not withstand the bending stresses. In other words, it would have succumbed to fatigue failure.

B. The Examiner's Burden of Proof in Citing Inherency

To establish a *prima facie* case of obviousness the Examiner must show some objective teaching in the art or that knowledge generally available to one of ordinary skill in the art would lead that individual to combine the teachings of the references. *In re Thrift*, 298 F.3d 1357, 1363 (Fed. Cir. 2002), citing *In re Fine*, 837 F.2d 1071, 1074 (Fed. Cir. 1988). When relying upon inherency, the Manual of Patent Examining Procedure §2112 specifically addresses what is required to make a *prima facie* case. As noted therein, the claiming of a new use, function or unknown property which is inherently present in the prior art does not necessarily make the claim patentable. MPEP §2112, citing, *In re Best*, 562 F.2d 1252, 1254 (C.C.P.A. 1977). However, this statement simply begs the question of what one must demonstrate to show that something is inherently present in the prior art.

A properly posited rejection based on inherency must meet a two-part test. First, the Examiner must show that the prior art product appears to be identical, except that it is silent on the inherent characteristic. This rationale applies to product, apparatus and process claims as to function, property or characteristic. Second, the Examiner must

provide a rationale or evidence tending to show inherency. MPEP §2112. As the MPEP notes:

> To establish inherency, the extrinsic evidence must make clear that the missing descriptive matter is necessarily present in the thing described in the reference, and that it would be so recognized by persons of ordinary skill. Inherency, however, may not be established by probabilities or possibilities. The mere fact that a certain thing may result from a given set of circumstances is not sufficient.

MPEP §2112, citing, *In re Robertson*, 169 F.3d 743, 745 (Fed. Cir. 1999). *See also*, *Akami Tech. Inc.,. v. Cable & Wireless Internet Svcs., Inc.*, 344 F. 3d 1186, 1192 (Fed. Cir. 2003), ("the dispositive question is whether one skilled in the art would reasonably understand or infer from the prior art reference's teaching that every claim [limitation] was disclosed in that single reference"), citing *Dayco Prods. Inc. v. Total Containment, Inc.*, 329 F.3d 1358, 1368 (Fed. Cir. 2003); *Crown Operations Intern. Ltd. v. Solutia, Inc.*,289 F.3d 1367, 1377 (Fed. Cir. 2002) ("if a limitation is inherently disclosed, it must necessarily be present and one of ordinary in the art would recognize its presence"). Moreover, "[i]n relying upon inherency the examiner must provide a basis in fact and/or technical reasoning to reasonably to support the determination that the allegedly inherent characteristic <u>necessarily</u> flows from the teachings of the prior art." *Ex parte Levy*, 17 U.S.P.Q.2d 1461, 1464 (Bd. Pat. App. & Inter. 1990).

Assuming the Examiner has demonstrated that it meets the two step test the burden then shifts to the applicant to demonstrate that the prior art reference does not necessarily or inherently possess the characteristics of the claimed product. *In re Peterson*, 315 F.3d 1325, 1330 (Fed. Cir. 2003).

C. <u>The Examiner's Application of the Allen Paper Fails to Meet the Requirements of the Two Step Test</u>

An examination of the Allen Paper in light of the two step test required to support an anticipation or obviousness rejection demonstrates that it <u>does not meet</u> this test. The first step is to determine whether it includes each and every limitation of the claimed invention but is silent on the claimed inherent characteristic. As noted above,

the Allen Paper discloses the smoothness ranges but is not silent on the claimed inherent characteristic. The Allen Paper teaches that the smooth cylinders in the claimed range and Reynolds numbers undergo significant VIV displacement. *See*, Allen Paper, page 683, col. 2, in its discussion of Figures 5 and 6. The Allen Paper discloses an apparatus having the same characteristics, but explicitly teaches away from the claimed inherent. Thus it cannot meet the first part of the test.

Further, since it teaches away from the claimed invention, it cannot be said that one of ordinary skill in the art would recognize the efficacy of an ultra-smooth surface to control both VIV and drag. One of ordinary skill in reading the Allen Paper would be lead to believe that the ultra-smooth would be insufficient to control both VIV and drag. Accordingly, the Allen Paper fails to meet the requirements of MPEP §2112 in determining inherency. Therefore, claims 1 and 4 are patentable over the Allen Paper.

D. The Examiner Ignored Objective Evidence on the Lack of Inherency

Even if the Examiner had made a *prima facie* case of obviousness based on inherency, the Examiner ignored objective evidence regarding the cited reference. In response to an Office Action mailed 20 March 2002, Attorney submitted a Response dated 27 August 2002. Attached to the response was the Declaration of Dr. Donald W. Allen (attached as Appendix C), co-author of the Allen Paper. In the Declaration, Dr. Allen addresses a number of concepts, including drag and VIV and discusses a number of references applied by the Examiner.

Dr. Allen specifically discusses the Allen Paper in paragraphs 10 – 13 of the Declaration. First, he discusses the K/D ratios of the ABS and PVC pipes used for the tests. He notes that the smoothness of the pipes was achieved by special surface preparation and rejects the notion that commercially available materials would have had the required K/D ratios. Allen Decl. ¶10. Dr. Allen then addresses the use of the strongback to prevent the test cylinders from vibrating to permit him to solely study drag. He specifically noted that "[w]ith the strongback inserted, the pipe was not free to move and did not undergo VIV." Allen Decl. ¶11. Thus, Fig. 4 with its reference to

strongback performance of the cylinders is irrelevant as the pipes were not free to by subject to VIV.

Dr. Allen then addressed the performance of the cylinders without a strongback and noted that both pipes underwent vibration and that drag increased as a result of the vibration. Specifically, it was stated that "significant VIV response is observed at supercritical Reynolds numbers and is accompanied by substantial increase in drag." Allen Decl. ¶12. In summary, he stated:

> There is nothing in the Allen article that suggests that an ultra-smooth surface as claimed in the present application is capable of reducing VIV in the ranges demonstrated. All flexible samples underwent significant VIV displacement over a broad range of Reynolds numbers.

Allen Decl. ¶13.

Dr. Allen made it clear that one of ordinary skill in the art would not reasonably understand or infer from the Allen Paper teaching that an ultra-smooth surface would result in a reduction in VIV and drag, as is required to support an assertion of inherency. *Akami Tech.*, 344 F.3d at 1192.

Despite this, the Examiner once again asserted inherency as a basis for rejection in the Final Office Action. The reasoning was that, despite the transitional use of the phrase in the claims, "consisting of" that the cylindrical elements could include additional structure in the nature of a strongback. This attempt to read additional structure into the claim despite the use of "consisting of" is clear error. *See*, *AFG Indus., Inc. v. Cardinal IG Co., Inc.*, 239 F.3d 1239, 1244 (Fed. Cir. 2001) ("closed" transition phrases such as "consisting of" are understood to exclude any elements, steps, or ingredients not specified in the claim), citing *PPG Indus. v. Guardian Indus. Corp.*, 156 F.3d 1351, 1354 (Fed.Cir.1998).

In light of the objective evidence presented in the Allen Declaration, even if the Examiner had made a *prima facie* case, it was rebutted and the burden was again on the Examiner. The Examiner's argument ignores the basic tenants of claim construction, the specification itself and focuses on an experimental element designed to eliminate VIV within the context of the test setup.

E. Claims 1 and 4 are Allowable Over the Cited Art

The sole basis for rejection of claims 1 and 4 is the Allen Paper and the assertion of inherency. As noted above, the Allen Paper does not meet the requirements of an assertion of inherency as it is not silent on the claimed inherent characteristic – it teaches the opposite. Moreover, one of ordinary skill would not recognize that smooth surfaces would decrease VIV response. The Allen Declaration makes it clear that the Allen Paper does not teach ultra-smooth surfaces for the purposes of VIV reduction. Accordingly, claims 1 and 4 are patentable over the cited art. Attorney requests that the Board direct an allowance as to these claims.

II. THE EXAMINER ERRED IN REJECTING CLAIMS 2, 3, 5 AND 6 AS OBVIOUS UNDER 35 U.S.C. §103(A) OVER THE ALLEN PAPER AND U.S. PATENT 4,470,722

In Paragraph 4 of the Final Office Action, the Examiner rejected claims 2, 3, 5 and 6 of the application under 35 U.S.C. §103(a) as obvious over the Allen Paper in view of Gregory '722 (Appendix D). With regard to claims 2 and 5, it was stated that the Allen Paper taught all the elements except that the ultra-smooth surface can be a coating. It was asserted that Gregory '722 taught in col. 4, lines 59 – 65 a cylindrical housing element for use iwith a marine production facility having an exterior coating of fiberglass or plastic. It was asserted that it would have been obvious to one of ordinary skill in the art to provide the ultra-smooth surfaces of the Allen Paper by means of a coating. With respect to claims 3 and 6, it was asserted that the Allen Paper taught all the elements except that the ultra-smooth surface can be a sleeve. The Examiner states that Gregory teaches in col. 2, lines 15 – 23 that fairings (sleeves) are commonly known to suppress VIV of a single riser. It was asserted that it would have been obvious to one of ordinary skill in the art to obtain an ultra-smooth surface by means of sleeves.

As noted in Section I of the Argument, the Allen Paper fails to teach that an ultra-smooth surface in the claimed range reduces VIV. If anything, the Allen Paper teaches

that VIV does occur on bodies having the required smoothness in the Reynolds number ranges, resulting in a significant increase in drag on bodies. Without this required underpinning, the addition of the Gregory '722 reference is insufficient. Gregory '722 does not overcome the lack of teaching that an ultra-smooth surface will result in a reduction in VIV. Gregory teaches placement of cylindrical housings lowered on guidelines about multiple subsea elements. It is asserted that by providing a single body, as opposed to multiple vertical elements that VIV and drag are reduced. There is little doubt that a system consisting of multiple vertical elements in a marine environment will result in increased drag and subjecting each of the elements to VIV (though U.S. Patent 6,148,751 teaches that by placement of circular bodies, one can break up vorticies). It is through the use of a sleeve over the multiple bodies to provide a single exterior shape that that allows one to predict the hydrodynamic forces on the system. Gregory '722, col. 6, lines 18 – 21. Despite the coatings disclosed in Gregory '722, there is nothing to suggest that the ultra-smooth surfaces in the claimed invention are taught by Gregory '722 or that they would result in a reduction in VIV response for the system.

The Allen Paper fails to teach that the ultra-smooth coatings of the claimed invention will result in a reduction of VIV and drag. Its combination with Gregory '722 does not address this lack of teaching. Nor does the fact that a coating and a sleeve is disclosed in Gregory '722 address this lack of teaching. Accordingly, claims 2, 3, 5 and 6 are patentable over the cited art. Attorney respectfully requests that the Board direct an allowance as to these claims.

II. CONCLUSION

Attorney has addressed each and every argument raised by the Examiner in the Final Office Action and has demonstrated how the prior art cited and arguments are deficient as a matter of law. Accordingly, Attorney respectfully requests that the Board instruct the United States Patent and Trademark Office to allow claims 1 – 6 and that this application proceed to allowance.

Respectfully submitted,

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APPENDIX A

1. A method of controlling drag and vortex induced vibration in a substantially cylindrical element consisting of providing an ultra-smooth surface about the cylindrical element having a K/D ratio of 1.0 X 10^{-4} or less where:

K is an average measure surface peak to through distance and

D is an effective outside diameter of the cylindrical element.

2. A method of controlling drag and vortex induced vibration about a substantially cylindrical marine element consisting of an ultra-smooth surface coating about the cylindrical element having a K/D ratio of 1.0 X 10^{-4} or less where:

K is an average measured surface peak to trough peak distance; and

D is an effective outside diameter of the cylindrical element including the coating.

3. A method of controlling drag and vortex induced vibration about a substantially cylindrical marine element consisting of an ultra-smooth surface on a substantially cylindrical sleeve about the cylindrical element having a K/D ratio of 1.0 X 10^{-4} or less where:

K is an average measured surface peak to trough peak distance; and

D is an effective outside diameter of the cylindrical element, including the sleeve.

4. A system for controlling drag and vortex induced vibration, consisting of:

a substantially cylindrical marine element have an ultra-smooth effective surface with a K/D roughness parameter of about 1.0 X 10^{-4} or less, where:

K is an average measured surface peak to trough peak distance; and

D is an effective outside diameter of the cylindrical element, including the sleeve.

5. A system for controlling drag and vortex induced vibration consisting of a substantially cylindrical marine element having an ultra-smooth coating material with a K/D roughness parameter of 1.0×10^{-4} or less where:

K is an average measured surface peak to trough peak distance; and

D is an effective outside diameter of the cylindrical element including the coating.

6. A system for controlling drag and vortex induced vibration consisting of a substantially cylindrical marine element having an ultra-smooth substantially

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Vortex-Induced Vibration Tests of a Flexible Smooth Cylinder at Supercritical Reynolds Numbers

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ABSTRACT

Tow tests have been conducted on flexible circular cylinders at supercritical Reynolds numbers in water. The tests were conducted at the Naval Surface Warfare Center's David Taylor Model Basin in Carderock, Maryland. Measurements were made of both drag and acceleration (due to vortex-induced vibration) of the cylinder.

An ABS pipe was used to achieve Reynolds numbers ranging from about 2×10^5 to 6×10^5, and a 5-9/16-in. diameter PVC pipe was used to achieve Reynolds numbers ranging from about 6×10^5 to 1.5×10^6. Tests were also conducted with aluminum inserts (strongbacks), made to fit just inside the test cylinders, in order to obtain stationary (rigid) cylinder drag measurements for comparison purposes. The test results for (relatively) smooth cylinders are presented in this paper.

KEY WORDS: vortex, vibration, riser, fatigue, Reynolds number, drag

INTRODUCTION

Vortex-induced vibration (VIV) of cylindrical structures is a fairly well-known phenomenon that can cause fatigue failure and/or excessive drag on a structure. In the ocean, many cylindrical structures are potentially subject to VIV from ocean currents. These structures include cables, risers, tendons, mooring lines and the hull of a spar-type structure.

One of the important parameters affecting VIV is known as the Reynolds number. The Reynolds number is defined as $Re = V*D/\nu$, where: V is the relative velocity of the flowing fluid that the cylinder experiences; D is the cylinder outside diameter, and ν is the kinematic viscosity of the fluid. Since the Reynolds number is proportional to both diameter and velocity, testing at Reynolds numbers that correspond to those of a prototype structure in which the velocity and diameter are often large (e.g., drilling risers with buoyancy and spar hulls in high currents), is quite difficult, especially in water where the associated forces are very large.

In the past, some researchers have performed high Reynolds number experiments by using air for the flowing fluid. Some of the more popular papers describing wind tunnel tests on stationary cylinders are those by Achenbach (1968), Jones et al. (1969), Roshko (1961), Schewe (1983), and Shih et al. (1992).

Reports of tests in water are more scarce. Rodenbusch and Gutierrez (1983) report on tests with a towed cylinder, not allowed to vibrate, that experienced Reynolds numbers over 2×10^6, however, suspected carriage vibrations prevented them from seeing a drop in drag coefficient for a smooth cylinder during the critical regime. (The critical Reynolds number regime is the flow regime in which the boundary layers become turbulent and is accompanied by a sharp reduction in the mean drag coefficient for a sufficiently smooth cylinder. This critical regime occurs at Reynolds numbers of about 1×10^5 - 1×10^5 and is dependent upon the cylinder's surface roughness and the free-stream turbulence of the flow. For purposes of this paper, Reynolds numbers above 1×10^5 are called critical, and those above 4×10^5 are called supercritical.) Some recent joint industry programs funded by oil companies have also produced drag measurements in water up to Reynolds numbers of 1.5×10^6, but the tests have been restricted to stationary cylinders or rigid (non-flexible) cylinders experiencing small forced oscillations (see also Jones et al. [1969] who forced a cylinder in air with small oscillations), and the information is not available in the public literature. To the authors' knowledge, while various field measurements have been made on cylinders such as marine risers experiencing currents (with all of the attendant inaccuracies of field measurements), controlled VIV measurements on flexible cylinders in water have been restricted to Reynolds numbers of about 350,000 (the latter are tests conducted by the author, but are unreported in the public literature) and are also quite scarce at these Reynolds numbers in air. This restriction is primarily due to the extremely large drag forces required to tow cylinders at high Reynolds numbers, thereby limiting the facilities that are capable of such tests (it is also possible to perform high Reynolds number tests in a wind tunnel or a large current tank such as the U.S. Navy's Large Cavitation Channel, but if the cylinder is allowed to vibrate, then there is a significant risk of propeller damage if the structural system should fail). Thus, controlled experimental information on VIV and the performance of suppression devices in water at supercritical Reynolds numbers is scarce. The objective of these experiments was to tow flexible and stationary (rigid) circular cylinders at speeds such that Reynolds numbers as high as possible could be achieved. A review of available

tow and current tank facilities in the U.S. led to the decision to perform tests at the David Taylor Model Basin (DTMB), where Reynolds numbers as high as 2×10^6 could be achieved. Both smooth and rough cylinders were towed, and several vortex suppression devices were also tested (note that the "smooth" cylinders used in these tests have some surface roughness associated with them as reported later in the paper). The results for smooth cylinders are presented in this paper. Results for the rough cylinders (which had much higher surface roughnesses than the smooth cylinders) and for the cylinders equipped with vortex suppression devices may be presented or published at a later date.

TEST DESCRIPTION

Model Basin

All of the experiments were conducted in the David Taylor Model Basin, which is part of the Naval Surface Warfare Center in Carderock, Maryland. For these tests, the high-speed basin was used. This basin has dimensions of 21 ft wide by 2968 ft long (of tow area), including a pneumatic wavemaker at one end and a wave-absorbing beach at the other end (see Figures 1 and 2). For the first 1169 ft, the depth is 10 ft and then is ramped down for 112 ft, so that the depth is 16 ft for the final 1687 ft. Models are towed such that the 10-ft depth section is encountered first. DTMB's carriage #5 was used to tow the cylinders. This carriage has a velocity range of 0 to 85 ft/sec and has a maximum power of 4800 hp. The carriage is rectangular in shape and is approximately 70 ft long by 26 ft wide. Eight weight-bearing vertical drive wheels and 4 opposed pairs of horizontal driving guide wheels operate in tandem on each side of the carriage. An open rectangular test bay 31.5 ft long by 10 ft wide was used for mounting the removable towing bridge to which the struts, which DTMB called "parachute struts", were attached. The struts were mounted to a frame and the entire frame was locked to the carriage on a set of four pads.

Test Setup

Two model pipe sizes were used for the tests (Figure 3 is a drawing of the test setup). The first cylinder was a 3-in. 145 psi ABS pipe with a 3½-in. outside diameter (D) and a 0.247 wall thickness. The other cylinder was 5-in. Schedule 40 PVC pipe with an outside diameter of 5-9/16 (5.5625) in. and a 0.258 wall thickness. The 3½-in. ABS pipe had a modulus of elasticity of 220 ksi (provided by the pipe manufacturer and consistent with the authors' previous tests on similar ABS pipes), and the 5-9/16-in. PVC pipe had a modulus of elasticity of 457 ksi (determined from tensile and compressive tests).

The fundamental natural frequency in bending for a smooth pipe was computed to be (using a finite element program) 5.77 Hz for the 3½-in. pipe and 8.83 Hz for the 5-9/16-in. pipe. Pluck tests were later conducted on smooth 5-9/16-in. pipes, and the average measured natural frequency was 8.48 Hz. The difference between the computed and measured values, as well as the differences between the individual pluck tests, were both attributable to variations in the pipe samples.

The average surface roughness of each cylinder was determined using a confocal scanning microscope and measuring the average peak-to-trough height of the topographical image of five samples. The average roughness for the PVC cylinder was $k/D = 9.94 \times 10^{-5}$ (this is the average peak-to-trough height divided by the cylinder diameter), with the samples ranging from $k/D = 8.86 \times 10^{-5}$ to 1.09×10^{-4}. For the ABS cylinder, the average roughness was $k/D = 1.37 \times 10^{-4}$, with the samples ranging from $k/D = 1.21 \times 10^{-4}$ to 1.51×10^{-4}.

All pipes were cut and faced to a length of 78.5 in. The ends were slightly counter-bored to fit aluminum end caps (Figure 3). The end caps were 0.5-in. thick with a 0.25-in. recess cut on the outside of the cap. The 0.25-in. recess allowed the end caps to be centered in the pipe. With the addition of the end caps, the total test pipe length was 79 in. Pins that were 1.25-in. long were attached to the end plates using 1.25-in. - 12 jam nuts. The outer end of the 1.25-in. pins were smooth. The smooth ends were mounted in the trailing load cell rod eyes (the total distance between the rod eye centers was 83.75 in.). This allowed the test pipe ends to float in the rod eyes. The ends were designed for movement (flexing) of the test pipe center (both horizontally and vertically) of ±15 degrees.

Two 10,000-lb load cells were used to measure drag loads. One load cell was mounted on each end of the test cylinder. The load cells were pinned to the load cell mounting plate in the front. The load cell mounting plates were bolted to the DTMB parachute struts with 3 bolts (1 in.) on each plate. Holes were available in each parachute strut to run the load cell and accelerometer lead wires from the test cylinder up to the carriage instrumentation area. The 1.25-in. male rod end at the rear of the load cell was captured in a steel housing. This housing allowed the load cell to measure loads in the towing direction. The housing also protected the load cell against side loads generated by the in and out motion of the vibrating test cylinders (the motions of the ends during bending motions). The 1.25-in. pins and end caps were held to the plastic test cylinders with a 92-in. by 0.5-in. B-7 all-thread rod, which ran through the center of the test cylinders. An accelerometer holder was mounted in the center of the all-thread and made to fit snugly inside the test cylinder with leaf springs. Two holders were fabricated for the 5-9/16-in. test cylinders and two holders for the 3½- in. test cylinders. For each pipe size, one holder was fabricated for the configuration having a strongback insert (aluminum pipe) and one was fabricated for the configuration without a strongback. The accelerometer holder was a 1.5-in. thick piece of aluminum round stock machined to mount leaf springs on its periphery. An accelerometer mounting hole was also machined in the aluminum round stock. The accelerometer's x-axis was mounted to measure accelerations in-line with the current (the tow direction), and the y-axis was mounted to measure the crossflow accelerations. The accelerometer was positioned at the center of the pipe span in order to measure vibrations at the anti-node of first-mode bending. The all-thread rod was torqued to approximately 75 in.-lbs for the 3½-in. test cylinders and 150 in.-lbs for the 5-9/16-in. test cylinders. The nuts were locked in place by double-nutting. A thick washer was used under the all-thread rod nut on the port side. This prevented the 1.25-in. pin from pulling out of the male rod eye in the event of a pipe failure. An anti-rotation arm was mounted on the starboard side of the test assembly, outside of the struts, to restrict torsional motions of the test cylinders without affecting the drag force measurements.

A number of tests were conducted using a "strongback" mounted inside the test cylinder. The strongback consisted of a piece of 78-in. long aluminum tubing with an outside diameter just smaller than the test cylinder (thus there were two strongbacks: one for a 3½-in. test cylinder and one for a 5-9/16-in. test cylinder) and a wall thickness of approximately 0.375 in. The function of the strongback was to restrain the test cylinder from experiencing vortex-induced vibrations so that stationary cylinder drag could be measured. The presence of the strongback accomplished this by increasing the natural frequency in bending to a value that is higher than expected to be excited during the tests. The aluminum end caps for the test configurations using a strongback were 0.75 in. thick. A 0.25-in. recess was used to center the strongback and a 0.25-in. recess was used to center the pipe.

Test Procedure

For each test, the carriage was first backed up well into the shallow portion (10-ft depth) of the tank (the carriage was usually backed up at a speed of 3 knots). Once the turbulence had subsided to an acceptable level (this usually required at least 5 minutes to reduce the turbulence to less than about 3 percent), a carriage run was executed (after zeroing the load cells). Data were taken from just before the initial carriage acceleration until the carriage reached final rest or had begun reversing direction for the next run.

For most tests, the carriage was accelerated at 1 ft/sec^2. Once the first targeted test speed was reached, the carriage operator would signal with a horn on the carriage. The carriage operator would again signal when the deeper portion of the tank was reached. At that point, the DTMB test engineer would insure that a minimum of another 300-350 ft of tank was traversed before signaling the carriage operator to accelerate to the next speed. Once this speed was reached, the carriage operator would again signal with the horn, and the DTMB test engineer would insure that another 300-350 ft of tank was traversed before signaling the carriage operator to end the test by decelerating and stopping the carriage. The 300-350-ft length of the tank required for each test (note that about 500 ft was actually traversed for most tests) was determined by the need for at least about 50 cycles of transverse VIV motion to make accurate measurements for each test. For example, at 30 ft/sec, the cylinder might vibrate as low as 8 Hz, but 300 ft would insure 10 seconds of data and thus 80 cycles (8 Hz times 10 seconds) of data. Since the last part of the data for each test speed was selected for analysis (discussed below), this allowed for some VIV cycles to build to a steady state before the analyzed portion of the data was obtained. Note that there are lighted distance markers on the side of the tank every 30 ft; this allowed for easy observation by the DTMB test engineer.

At a speed of approximately 13.4 knots, the Froude number (with depth as the length scale) is unity. Degradation of the measurements was a concern for tests near this speed, however, examination of the data obtained in these experiments found the degradations (these degradations were estimated based upon the measurements of the surrounding data points and looking for changes at this tow speed, e.g., dips in the drag coefficients, etc.), if present at all, to be quite small for the transverse displacements and the drag force measurements (less than 3 percent), but substantial for the in-line displacement measurements (see discussion below in the section on Test Results). There was often significant spray and splashing near this speed.

Instrumentation and Data Acquisition

Tow speed measurements were made by feeding the carriage speed measurements (made by a magnetic pick-up on one of the drive wheels) directly into the data acquisition hardware. The excessive drag on the carriage sometimes caused, for the 5-9/16-in. cylinder tests, a slight reduction in the velocity that was preset by the carriage control system (due to some problems with the feedback loop). This was on the order of 1 to 1.5 percent. However, it should be noted that it is the actual tow velocity that is recorded with the results, since this speed reduction resulted in a reduction of the wheel r.p.m.

Turbulence was measured by a hot wire anemometer, with the results output to a strip chart recorder. These measurements were used to determine the time needed between tests for the turbulence to decay to an acceptable level (an intensity of about 3 percent or less).

The in-line drag forces were measured using custom load cells built by Houston Scientific Intl. (Model No. 3100-0010) with a range of 0 to 10,000 lb. Sixty feet of marine-type bronze armored cable was sealed to each cell with a 04-001 submersible seal. UNF mounting threads (1.25 - 12) were selected, which allowed the ends of the load cells to be fitted with matching male rod eyes (1.25 - 12). The load cells were each calibrated and checked for linearity prior to testing.

A calibration of the entire test assembly was conducted at Shell's Gasmer Road facility in Houston prior to shipping to DTMB. This assembly consisted of everything below the struts, including the end plates and a 5-9/16-in. PVC test cylinder. A 6-in. steel pipe was used to simulate the strut frame, and loads were applied to the PVC test cylinder by suspending custom-made steel canisters of sand from the vertically mounted test assembly (i.e., the assembly was rotated 90 degrees and loaded vertically instead of horizontally). A total of four canisters were used, and both smooth and rough pipes were loaded until yield (which occurred at around 3000-3200 lb). The load cells at the cylinder ends were compared with a load cell that measured the suspended weight of the assembly (this assembly load cell was zeroed with the unloaded assembly weight prior to the test so that it would measure the applied loads on the PVC pipe). The test results showed that the two cylinder load cells (one at each end) were within 1.5 percent of each other and that their combined readings were within 0.5 percent of the assembly load cell reading throughout the load range. Also, a pull test in air with the assembly mounted to the struts was later performed at DTMB with a load up to 1000 lb. and similar results were observed.

The water viscosity was determined by performing several temperature measurements during three days of the tests and using well-known values for the viscosity of freshwater at a given temperature. Since the measured value of the water temperature was 69.5 degrees Fahrenheit, the kinematic viscosity value used was 1.06×10^{-5} ft^2/sec. This value was also checked by taking a water sample and performing measurements using a viscometer at 69.5 degrees Fahrenheit.

Columbia HEVP-14 biaxial accelerometers were used to measure the accelerations. One accelerometer was mounted inside the accelerometer holder at the center of the test cylinder; a second accelerometer was located inside the bottom of the starboard strut; and a third accelerometer was located on the top of the strut frame on the bridge. Prior to the tests, the accelerometers were calibrated with a shaker assembly and were found to be accurate to within 1.7 percent over the range of 10 to 20 Hz.

The analog accelerometer voltage signals were first amplified using Columbia Model 9002 charge amplifiers. The amplified signals were then digitized by LABTECH NOTEBOOK data acquisition software, controlling a Data Translation DT2801 board, and stored on the hard disk drive of a Winbook laptop computer. The signals for the 5-9/16-in. cylinder tests were also stored in analog form by a TEAC XR510 tape recorder. The analog signals were low-pass filtered at 40 Hz (to eliminate tank noise, electrical noise, etc.) for the 3½-in. cylinder tests and at 50 Hz for the 5-9/16-in. cylinder tests, using Frequency Devices analog filters, before the data were acquired by LABTECH NOTEBOOK.

Data Analysis

Since two tests (i.e., two velocities) were usually performed for each carriage run, a stripping program was written for stripping out the best data for each test. This was accomplished by finding the range of data points with measured velocities that matched the test velocity and then using the last 2048 points. By using the last 2048 points as opposed to the first 2048 points, transients were best eliminated and, for the first test of each carriage run, the data from the 16-ft depth were

used instead of any data that were taken during the 10-ft depth used for the approach.

The data analysis was performed using a custom data analysis package on an HP730 workstation, which performed the following steps for each test:

1) The raw accelerations were scaled according to the gain settings on the charge amplifiers and converted to the proper engineering units;
2) The accelerations were next Fast Fourier Transformed to produce frequency spectra of both the in-line and transverse directions;
3) Acceleration power spectra were computed from the frequency spectra for both the in-line and transverse directions.
4) The acceleration power spectra were filtered from a high-pass filter setting to a low-pass filter setting. The low-pass filter setting was 30 Hz for the 3½-in. test cylinder and 40 Hz for the 5-9/16-in. test cylinder. The high-pass filter setting was varied for each test in order to eliminate low frequency noise from the displacements resulting from double integration of the accelerations. This high-pass filter setting was chosen based upon a review of the power spectra and was an iterative process, but was usually about 3 Hz (high-pass filtering is required in order to obtain displacements since the frequency squared is in the denominator). A 10-point cosine taper was used in the filtering process.
5) The acceleration power spectra were twice integrated to compute rms displacements for that test.
6) The mean drag coefficients were computed from the mean drag forces (arithmetic average) that were measured by the in-line load cells.

Note that during testing, the above steps were performed to produce a preliminary determination of data validity using the Winbook computer.

The mean drag coefficients were computed from the formula $C_d = F_d/(0.5*\rho*V^2*D*L)$, where: C_d is the drag coefficient; F_d is the drag force; ρ is the density of the water in the tank (the specific weight of 62.44 lb/ft^3 divided by the gravitational constant); V is the tow velocity; D is the cylinder outside diameter; and L is the cylinder length of 79 in. (includes end plates).

TEST RESULTS

The following test sections present the test results. For simplicity, the results are plotted as a function of Reynolds number, however they could have been plotted against tow velocity or reduced velocity instead. Thus, it is important to note that there are parameters other than the Reynolds number that also influence the results.

For purposes of the following discussions, the subcritical Reynolds number range is defined as consisting of Reynolds numbers below 1×10^5; the critical Reynolds number range will be defined as approximately consisting of Reynolds numbers of 1×10^5 to 4×10^5, and the supercritical range defined as approximately 4×10^5 to 3.5×10^6 (the transcritical range consists of Reynolds numbers above 3.5×10^6).

Stationary Cylinder Results

Figure 4 is a plot of the drag coefficients, together with data from other researchers, for the tests with a strongback (aluminum pipe insert used to keep the pipe from vibrating). This figure shows a steep decrease in the drag coefficient through the critical Reynolds number range, known as the "drag crisis." The drag coefficient then continues to slightly decrease as the Reynolds number is increased through the supercritical range. Whereas the drag coefficients for the 3½-in. stationary cylinder were decreasing slightly up to a Reynolds number of just over 6×10^5 (the highest Reynolds numbers tested with this cylinder), the drag coefficients for the 5-9/16-in. cylinder show an increase for Reynolds numbers beginning at 8.8×10^5. Figure 4 indicates that the Reynolds number at which the drag coefficient for a stationary smooth cylinder reaches a minimum in, the supercritical range varies from about 3×10^5 to about 8×10^5. Thus, the data herein would appear to support a minimum in the upper end of this range. The differences between the present data and the data from most of the included resources in Figure 4 are most probably related to surface roughness, as is consistent in comparing the present data with the data from Shih et al (1992). However, other parameters such as free-stream turbulence and cylinder vibration can also have an effect (for these tests the turbulence intensity was less than about 3 percent and the cylinder in-line and transverse rms displacements were less than 0.0024D, as measured by the pipe accelerometer).

Vibrating Cylinder Results

Figure 5 is a plot of the results for the smooth ABS cylinder without a strongback (thus the cylinder is free to experience VIV). The substantial VIV resulted in a significant increase in the drag coefficients (about 50-80 percent). These results illustrate that significant VIV can, and does, occur at critical and supercritical Reynolds numbers and that the corresponding increase in drag coefficient is similar to that of subcritical Reynolds numbers.

Figure 6 is a plot of the results for the smooth PVC cylinder without a strongback, and shows both substantial vibration and significant increased drag due to the vibrations. As with many of the tests with this pipe diameter, the tests were limited by what the PVC pipe could structurally withstand, thus the velocities were not increased until the pipe "locked-out" (i.e., there is a velocity at which the first-mode VIV of the pipe will cease due to sufficient differences between the vortex shedding frequency and the natural frequency, but the pipe could not withstand the bending stresses associated with velocities this high).

Figure 6 shows one data point at which the in-line displacement was larger than that for surrounding data points. This test speed (24 ft/sec) consistently produced this phenomenon, and it should be noted that it occurs just above the speed at which the basin Froude number is unity. However, tests with a cylinder having a different diameter, or tests with a different facility are needed to determine whether this is a Froude number or a Reynolds number effect. Note that at this test speed (about 24 ft/sec), the reduced velocity, based upon the measured natural frequency of 8.48 Hz, is about 6.1. This is the approximate reduced velocity at which others have observed a phase change between the cylinder motions and the forces on the cylinder (e.g., see Wu [1989]). Thus, this phase change may also play a role in the large in-line motions at this test speed.

Figure 7 contains plots of the acceleration and displacement power spectra for the test with a Reynolds number of 1.39×10^6 (test speed of 31.7 ft/sec). The large in-line peak corresponding with the transverse (cross-flow) peak at about 8.5 Hz indicates that the cylinder is vibrating at an angle of about 0.6 degrees from the cross-flow direction. This is most likely due to the normal misalignment of the accelerometer and not due to actual cylinder vibration. The small magnitude of this misalignment is indicative of the high accuracy of these measurements.

SUMMARY AND CONCLUSIONS

Tests have been conducted on flexible and stationary cylinders in water at Reynolds numbers over 1.5×10^6. The results indicate that:

a) small levels of roughness can have a tremendous effect on the drag coefficient of supercritical Reynolds number flow past stationary and vibrating cylinders; and

b) significant VIV response is observed at supercritical Reynolds numbers and is accompanied by substantial increases in the mean drag coefficient.

These tests indicate that future test programs on cylinders experiencing flow at supercritical Reynolds numbers should carefully measure and report the values of surface roughness (in addition to other relevant parameters such as turbulence intensity), even for a cylinder thought to be "smooth." Careful studies of the effects of Reynolds number, surface roughness, and turbulence on the VIV response and drag of circular cylinders may allow more accurate determination of the response of a variety of cylindrical offshore structures exposed to high Reynolds number flows.

ACKNOWLEDGMENTS

A research project as demanding and ambitious as this one required tremendous support from a group of over 40 people who directly contributed to this project. The authors are especially grateful to Doug McMullen, who provided data analysis and testing support, Joe Haws, who provided data acquisition support, Dave McMillan, who provided drafting support, and Robert Patterson, who, with Shell E&P Technology Co., provided administrative and financial support. The authors are also especially appreciative of the staff at the Naval Surface Warfare Center - Towed Systems Branch who oversee the DTMB and without whom these tests would not have been possible.

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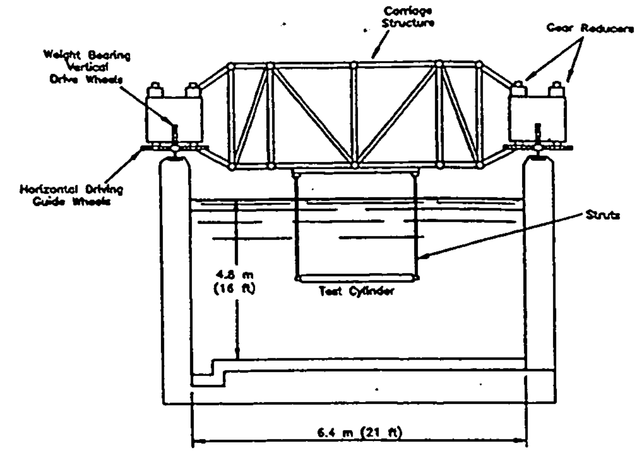

Fig. 1 Elevation View of Deep End of High Speed Basin and Carriage Number 5

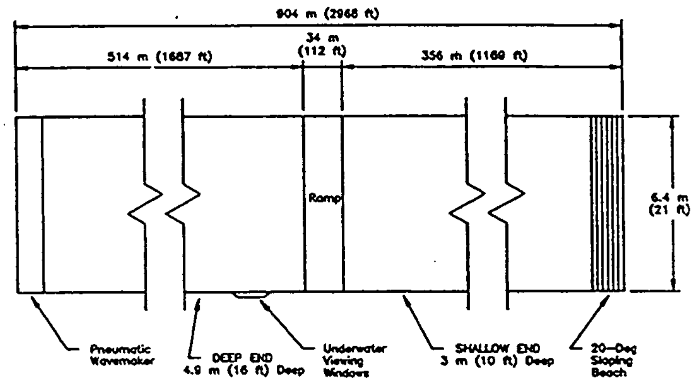

Fig. 2 Schematic Plan View of High Speed Basin

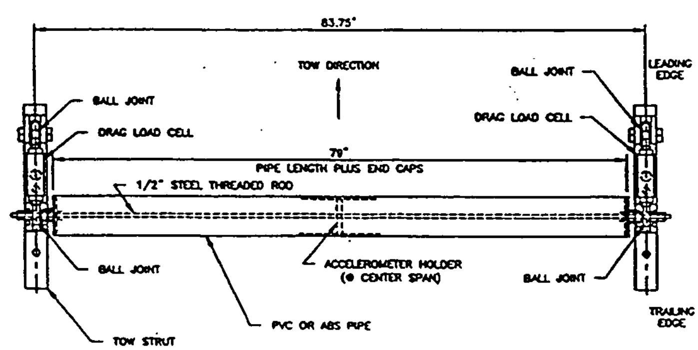

Fig. 3 Test Setup (Bottom View Looking Up)

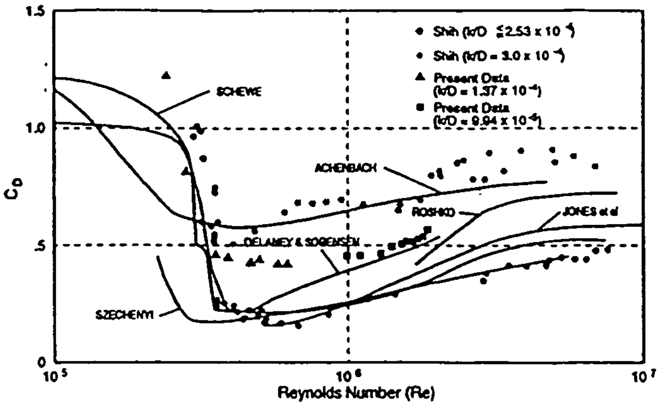

Fig. 4 Stationary Cylinder Drag Coefficients

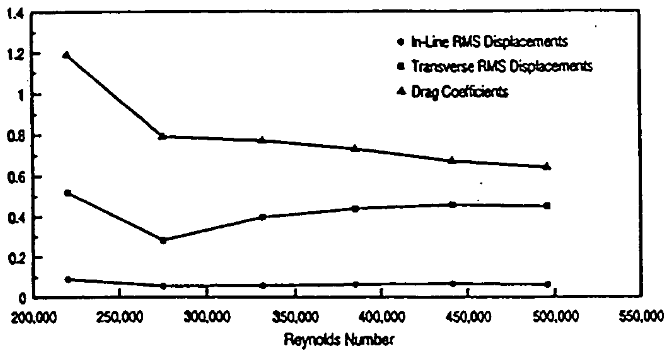

Fig. 5 Displacement and Drag Coefficient Results for 3 1/2 in. Diameter Smooth Cylinder Tests

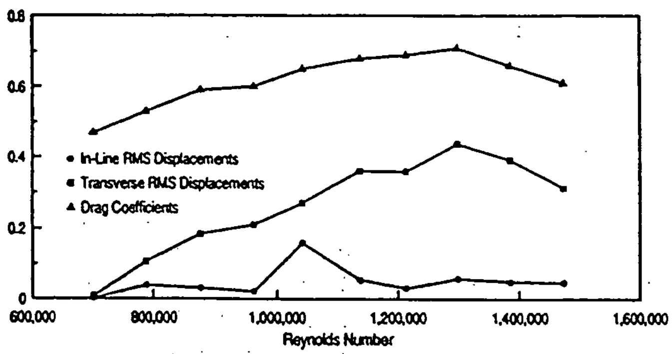

Fig. 6 Displacement and Drag Coefficient Results for 5 9/16 in. Diameter Smooth Cylinder Tests

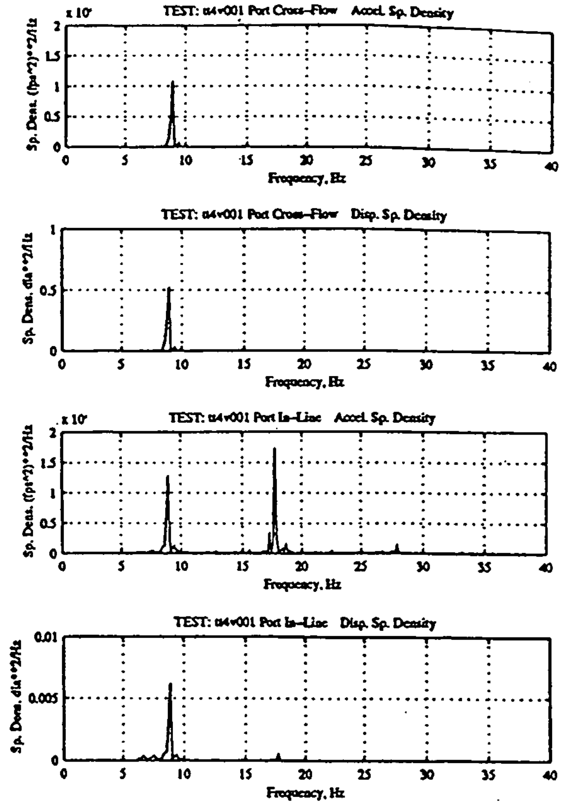

Fig. 7 Acceleration and Displacement Power Spectra

IN THE UNITED STATES PATENT AND TRADEMARK OFFICE

In re application of Donald W. Allen et al Serial No. 09/625,893 Filed: July 26, 2000 SMOOTH SLEEVES FOR DRAG AND VIV REDUCTION OF CYLINDRICAL STRUCTURES	Group Art Unit: 3673 Examiner: K. Mitchell August 22, 2002

DECLARATION OF DR. DONALD W. ALLEN

I, Dr. Donald Wayne Allen declare as follows:

1. My name is Dr. Donald Wayne Allen. I am the Vortex-Induced Vibration and Suppression Team Leader for Shell Global Solutions (U.S.) and named inventor on the above-referenced patent application. I am more than 18 years of age, have not been convicted of a felony or a crime of moral turpitude, am of sound mind, and am competent to make this Declaration.

2. I received a B. S. in Mechanical Engineering from Texas A&M University in 1981, and a Ph. D. in Mechanical Engineering from Rice University in 1986. I have worked for Shell since August 1986 and am a Senior Staff Research Engineer. I consult with various Shell and non-Shell entities regard the potential vortex-induced vibration ("VIV") problems with various subsea structures. In this role, I have performed VIV analyses of offshore and subsea structures such as production platforms, risers, riser bases, jumpers, tendons, spars, and pipeline spans.

3. I have performed research directed to the characterization of VIV response prediction. I have also performed research directed to the development of VIV suppression devices, including various helical strake systems, fairing systems and shroud or covering systems. This research includes work performed at various Shell facilities located in Houston, Texas and at the Naval Surface Warfare Center, located in Caderock, Maryland.

4. I have authored and published a number of papers on the subject of VIV and its suppression. A list of my publications is attached hereto as Exhibit 1.

5. I have reviewed the Office Action that was issued in the above-referenced application and believe that the Examiner is in error as to (a) various hydrodynamic problems addressed and (b) what the references teach.

6. Laminar flow, for an incompressible fluid such as water or air at low Mach numbers, may be defined as the streamline flow of a fluid in which the fluid moves in layers without fluctuations or turbulence so that successive particles passing the same point have the same velocity. Laminar flow about a body typically takes place at low fluid velocities, high viscosities, low densities or small body dimensions. Conversely, turbulent flow is a form of fluid flow in which the particles of the fluid move in a disordered manner in irregular paths, resulting in an exchange of momentum from one portion of a fluid to another. For flow past a an object with a round, rectangular, or otherwise bluff cross section, turbulent flow takes over from laminar flow when high values of Reynolds number are reached. The Reynolds number itself a dimensionless parameter and may be defined as:

$$R_e = \frac{\rho Ul}{\mu} = \frac{Ul}{\nu}$$

where ρ is the fluid density, μ is the dynamic viscosity coefficient, ν is the kinematic viscosity coefficient (μ/ρ), U is the characteristic velocity of the fluid and L is a length parameter, such as the diameter of a pipe or diameter or length of a body, in the case of flow external to a body. As the Reynolds number increases, the flow past a body becomes increasingly turbulent. In the hydrodynamic context, drag is the force exerted on a body in a direction parallel to the fluid flow.

7. Vortex shedding is a phenomenon that occurs in flow past a cylinder. As the fluid approaches the cylinder, the fluid closest to the cylinder wall is impeded by the friction on the cylinder surface, hence the region between the wall and where the velocity is close to that of the free stream is known as the boundary layer. As the boundary layer fluid proceeds around the cylindrical surface, the friction of the surface causes the boundary to flow increasingly slower, eventually bringing it to a halt. The boundary layer then separates from the cylinder, forming a shear layer. Because the fluid flowing in outer portion of the shear layer is flowing faster than the inner portion of the layer, the layer rolls up into a vortex. This process of separation and formation of vorticies can occur in both the laminar and turbulent flow regimes.

8. As the Reynolds number increases, the vorticies become increasingly unstable and start separating or shedding in close proximity to the cylindrical body. When this occurs, an alternating vortex or von Karman street is formed. Where the cylindrical body is free to move, the alternating forces will cause the cylinder to vibrate. If the cylinder vibrates at its natural frequency it will tend to lock into this vibration mode. If the mode shape is a bending mode (typcial of a long slender tubular such as a riser or pipeline) then the cylinder will see increased bending stress, which can induce fatigue in the body, thereby decreasing its serviceable life.

9. In reviewing the Office Action, I further noted that in many instances, the Examiner correctly noted that a smoother surface can result in a reduction in friction of fluid flowing around a body, thereby reducing drag. The Examiner reaches the conclusion that a

reduction in drag will result in a reduction in VIV. VIV, while related to, differs from drag. While it is true that an increase in VIV can result in an increase in drag, it does not necessarily follow that a decrease in drag results in a decrease in VIV. It is known that certain structures that increase drag on a body can result in a decrease in VIV. The application of helical strakes or fins on cylindrical bodies results in an increase in drag. However, the helical strakes have the effect of breaking up vortex formation, thus decreasing VIV. In US Patent 6,309,141, in which I am named a co-inventor, the fact that an increase in drag can, in certain circumstances, result in reduced VIV, is reported at column 3, lines 18 – 27. Thus, it does not follow that a reduction in drag necessarily results in a reduction in VIV.

10. I have reviewed the Examiner's rejections and the prior art relied upon for the rejections. The Examiner states that U.S. Patent 4,470,722 to Gregory ("Gregory '722") teaches a cylindrical housing element for use with a marine production facility that has an exterior coating of fiberglass or plastic. The Examiner then states that while Gregory does not explicitly disclose an ultra-smooth cylinder, the fact that the present application states that the ultra-smooth surface could be provided by sleeves made of copper, carbon fiber, rubber or any sufficiently smooth material. The present application discloses an example of an ultra-smooth surface as having a K/D of 5.1×10^{-5}. The Examiner then argues that since Gregory discloses fiberglass, it must inherently have a K/D of 5.1×10^{-5} or less.

11. This assumption is in error, as it does not follow that fiberglass or plastic has a K/D of the type claimed in the present application. In the article *Vortex-Induced Vibration Tests of a Flexible Smooth Cylinder at Supercritical Reynolds Numbers*, May 1997, (the "Allen article") I discussed tests utilizing ABS or PVC plastic bodies. The K/D ratios discussed therein were in the range between 8.86×10^{-5} to 1.51×10^{-4}. Both the ABS and PVC cylinders required special surface preparation prior to the tests. In this instance, the ABS pipe was purchased commercially, chosen from numerous samples for its surface smoothness (i.e. lack of scratches and scuffs that typically accompany commercially purchased plastic pipe or tubing), and specially wrapped for transport to realize the K/D ratio set forth in the article. Similarly, the PVC was purchased commercially, chosen out of numerous samples for its surface smoothness, specially wrapped for transport purposes, and then wiped with an acetone to realize the smoothness described in the article. Contrary to the Examiner's statement, it does not follow that because Gregory '722 discloses fiberglass that the fiberglass would have the K/D of the claimed invention. In fact, in the experiments conducted in which this discovery was made, a fiberglass pipe without any modifications to its surface sustained very high VIV. The discovery was made when a fiberglass pipe was surface ground to an extremely smooth finish. Only then did was the VIV greatly reduced.

12. The Examiner also rejected claims 1 and 4 based on the Allen article stating that the Allen article teaches a method and system for controlling drag and VIV. Specifically at page 683 with reference to Fig. 4, the discussion was related to stationary cylinders in which a "strongback" was inserted in order to keep the pipe from vibrating. With the strongback inserted, the pipe was not free to move and did not undergo VIV. The tests dealing with the strongback pipe studied the relationship between the Reynolds numbers and the drag coefficient. It was posited that the surface smoothness of the pipe accounted for the

differences between the pipe being studied and data from existing references which were also for stationary cylinders.

13. The Allen article talks about VIV beginning at page 683, second column, first full paragraph. Therein, the ABS cylinder, with k/D in the range of $1.21 - 1.51 \times 10^{-4}$ (see page 681, col. 1, last full paragraph) was tested without a strongback and was free to vibrate. As noted, the VIV increased, as did drag. In Fig. 6, the smooth PVC cylinder, having a k/D in the range of 8.86×10^{-5} to 1.09×10^{-4} was tested without a strongback. As noted therein, it displayed "substantial vibration and significant increased drag due to the vibrations." The conclusions are that for flexible and stationary cylinders in water with an Re over 1.5×10^{6}, the results indicate that

> a) small levels of roughness can have a tremendous effect on the drag coefficient of supercritical Reynolds number flow; and
> b) significant VIV response is observed at supercritical Reynolds numbers and is accompanied by substantial increase in drag.

14. There is nothing in the Allen article that suggests that an ultra-smooth surface as claimed in the present application is capable of reducing VIV in the ranges demonstrated. All flexible samples underwent significant VIV displacement over a broad range of Reynolds numbers.

15. The Examiner also cited U.S. Patent 6,206,614 and articles by Sellens and Smith are grounds for rejecting the claims of the patent. The Blevins '614 patent teaches a cylindrical sleeve and a method for controlling VIV by means of spacing of the columns. (Col. 4, lines 13 – 45). The Examiner then goes on to state that the Sellens article, the CE/ME 101 handout and the Smith article teach that smooth surfaces create less turbulent flow. It is also stated that the definition of relative roughness appearing at page 2 of the Sellens article is equivalent to the K/D ratio described in the present application and relates this to the drag coefficient, Reynolds number and relative roughness. I do not disagree. This is work that was similar to that discussed with reference to Fig. 4 of the Allen article.

16. While the Sellens and Allen articles do discuss surface smoothness and its effect on the drag coefficient over a range of Reynolds numbers, neither article suggests or teaches that surface smoothness has a significant effect on VIV suppression. As noted above, in discussion of Fig. 5 (pipe free to vibrate), the sample saw a significant VIV response in terms of displacement over a broad range of Reynolds numbers.

17. The CE/ME 101 handout is cited as teaching that relative roughness increases the turbulent drag and flow over cylinders. The Van Dyke reference merely identifies the "[e]ffect of rough surface or turbulent free stream" as opposed to that portion of the graph relating to turbulent boundary layers. While there is a depiction of a von Karman vortex street at Re = 140, there is no discussion with respect how varying degrees of surface roughness might suppress VIV (interestingly enough, at Re = 140 the flow past a cylinder is in fact completely laminar). Similarly, the *Drag of Blunt Bodies and Streamlined*

Bodies and the email from Professor Smits all address the very same issue – the reduction of drag over a range of Reynolds numbers can be a function of the smoothness of the surface. None of them address a reduction in VIV as a function of smoothness.

18. While it is true that a significant VIV response will increase the drag seen by a cylindrical body (see Allen article at 684), it does not follow that a reduction in drag will necessarily result in suppression of VIV. The Allen article, Mech 441, CE/ME 101, *Drag of Blunt Bodies and Streamlined Bodies* and Smits article do not suggest that an ultra-smooth surface can operate to significantly decrease VIV as claimed in the present invention.

I am aware that willful false statements and the like are punishable by fine or imprisonment, or both under Title 18 U.S.C. §1001 and may jeopardize the validity of the application or any patent issuing hereon. All statements made herein are made based on my own knowledge are true and that all statements made on information and belief are believed to be true.

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D. Allen

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